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By

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ABSTRACT

Mathematical models of the heat transfer in cooled gas-turbine blades have little empirical justification. If the flow phenomena that influence heat transfer could be photographed more accurate estimates of the heat transferred in a cooled gas turbine could be made.

In view of the high temperatures and the expense of operating a gas turbine, the first stage of a cooled gas turbine was operated in a water tunnel. Since only the inlet guide vanes and the first stage rotor were available, a support assembly had to be designed and constructed to hold the turbine components and sufficient flow had to be provided to turn the rotor.

The open-channel water tunnel in the Hydrodynamics Laboratory was modified in order that a test section large enough to accommodate the gas turbine could be installed. A support assembly was designed and constructed for the turbine which provided adequate radial and axial support. Incorporated in the support structure were a pressurized grease system, independent coolant (dye) systems, and contraction surfaces to accelerate the flow through the turbine blades.

Upon completion of the assembly and of the installation of the test section, a series of thirteen testing runs were conducted. After the third test, significant repairs had to be made to the accelerating surfaces. During final series of tests photographs were taken of the dye emerging from the rotor blade tips.

The quality of the photographs was vitiated by insufficient lighting, parallax, and refraction. Furthermore it was evident that operating conditions conducive to flow visualization would be far from Reynolds similarity.

Efforts will have to be initiated to refine the photographic techniques employed if satisfactory resolution and definition is to be attained. If these obstacles could be overcomed, the results would enable designer to develop

more effective gas-turbine cooling systems.

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Title: Associate Professor of Mechanical Engineering

ACKNOWLEDGEMENTS

I would like to thank Philip Haley for the significant contribution he made to the design and construction of the test section and subsequently to the testing of the gas turbine. Phil intends to continue on with the work begun in this experiment and hopefully attain more meaningful results. I wish him the best of luck.

I would like also to thank Dr. Harold Edgerton for his timely advice in the area of underwater photography. His assistance obviated many hours of research in this field by the author.

Finally, I wish to cite my wife, Eileen, for her patience and perseverance while typing this paper. Construction setbacks prevented an early completion; however she was able to convert a rather rough draft into a finished product in very short order.

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1. INTRODUCTION

Materials currently employed in the manufacture of gas-turbine blades limit turbine-inlet temperatures to values below 2000 °F unless some form of blade cooling is used. Since the maximum efficiency and specific power of gas turbines increase with increasing inlet gas temperature, advances in uncooled turbine ratings are dependent on devoloping better metals. If gas turbines could operate for long periods at temperatures over 2000 °F, the performance, i.e. weight per shaft horsepower and efficiency could be improved substantially.

An alternative approach to high-temperature design which eases the metallurgical problem is to cool gas-turbine blades. In such a system, turbine-inlet temperature may be raised several hundred degrees over blade temperature. Brown [1] showed that an increase in T.I.T. from 1250 °F to 2200 °F in a typical gas turbine with intercooling and heat exchanger but no reheat would raise the efficiency from 33.5 to 46.0 per cent and reduce the specific mass flow from 62 to 22 lb/hp h.

These results have generated great interest in the investigation of blade-cooling techniques. If an optimum cooling system is to be developed, a thorough knowledge of heat transfer through the blades and an understanding of the phenomena that influence it are essential. Ainley [2] has shown that there is a significant difference in the local heat-transfer coefficient obtained in a stationary cascade [3] and the distribution found in an operating turbine. Apparently, secondary flows produced by non-uniform conditions in the flow approaching the turbine blades, stream turbulence, and the centrifugal effect on the boundary layer produce this change in heat transfer [4]. If heat removal is to be optimized through the blade

rows, the magitude, extent, and location of secondary flows must be determined. Furthermore, an understanding of the effect of cooling flows on secondary vortices, boundary-layer growth, separation, and turbulence would facilitate the design of more efficient cooling systems.

Previous investigations of secondary flows have been limited principally to stationary cascades. This work is discussed in Appendix A. Several attempts have been made, e.g. reference (18) to visualize secondary-flow phenomena in radial compressors. However the interaction of discharge cooling flows with normal turbine flow regimes has not been photographed in a stationary or a rotating medium. The results desired from the present research are firstly a qualitative determination of the flow patterns which occur in different circumstances and a consequent improvement in the theoretical models on which analytical heat-transfer predictions can be based.

II. Procedure

The visualization of flow patterns in an operating gas turbine was virtually impossible due to the high temperature experienced. However if Reynolds numbers were conserved, full momentum similarity could be achieved. Thus it was decided to operate the gas turbine in a water rig. The large increase in viscosity would cause a substantial reduction in velocity, and thus a condition more conducive to flow visualization could be investigated.

Initially a facility which was compatible with the requirements of the experiment, i.e. flow rate and turbine size had to be located and constructed. Since only the first-stage rotor and stator were available for testing, a support system had to be designed and constructed. Additionally a casing for the gas turbine had to be manufactured. The casing had to be approximately the same diameter as the inlet guide vanes.

If flow regimes were to be photographed, it was imperative that a uniform velocity distribution entered the inlet guide vanes. This could be achieved by accelerating the flow through an annular contraction.

The ability to photograph the egressing coolant flows had to be incorporated into the design. There were several areas of interest; however areas between the blade would require the use of fiber-optic bundles. In view of problems that would be encountered in providing an access for the bundle, only the dye traces exiting the rotor were photographed.

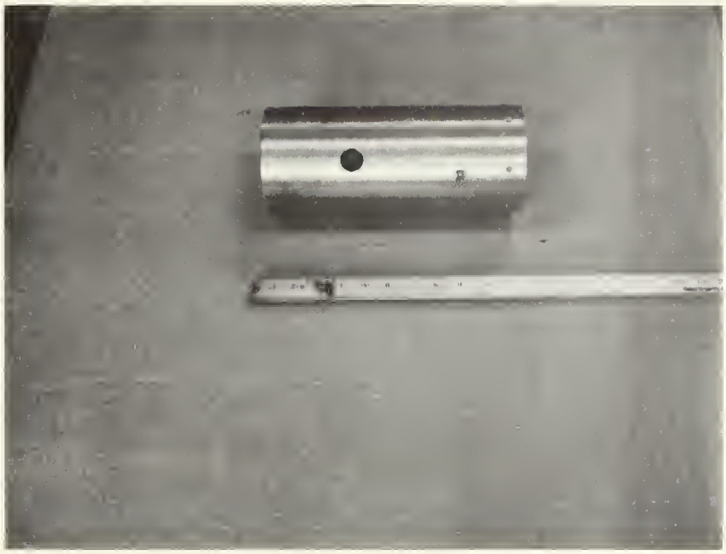
The experimental work was divided into two major phases, test-section construction and data collection.

Initially the original open-channel water tunnel was dismantled and dimensions were determined for the design of the new test section. The test section was fabricated

by an outside contractor while modifications were underway on the water tunnel. An " A " frame was built to support the diffuser. The new support and the original steel frame were relocated and the diffuser was mounted on the supports to establish proper alignment with the contraction section. The mating flanges were drilled in placed and the two sections were joined by 8 5/8-inch bolts and a 3/8-inch rubber gasket. This completed the modification of the water tunnel (see figure B-4).

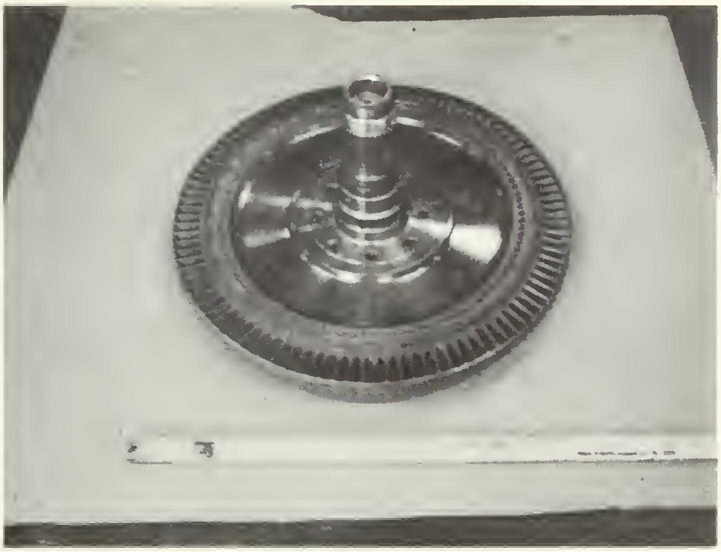
Due to long disuse, the water tunnel required some maintenance prior to testing. The 90% elbow on the discharge pipe was removed and the plate valve on the pipe (see figure B-2) was inspected. The lifting mechanism for the plate valve was freed and lubricated. The valve was cycled several times to ensure proper operation and to verify the accuracy of the valve-position indicator. The filter system which consisted of four screens mounted on steel frames were removed and cleaned. Accessible debris contiguous to the filter box was removed at this time. Other areas readily accessible which experienced corrosion were cleaned. All original components were reassembled in preparation for the tests.

Next, the rotor / stator mounting assembly was built. The bearing housing, O-ring supports, bearing spacers, and spline sleeve were machined in accordance with design specifications. The outer races, bearing spacer, and retainer rings were pressed into the bearing housing. The upstream inner race and O-ring support were pressed on to the spline sleeve. The sub-assembly was inserted into the bearing-housing assembly. The inlet guide vanes were mounted on to guide-vane support plate and secured with hose clamps. The support plate was mounted on to the rotor. The downstream O-ring support and bearing inner race were pressed on to the rotor. The housing assembly



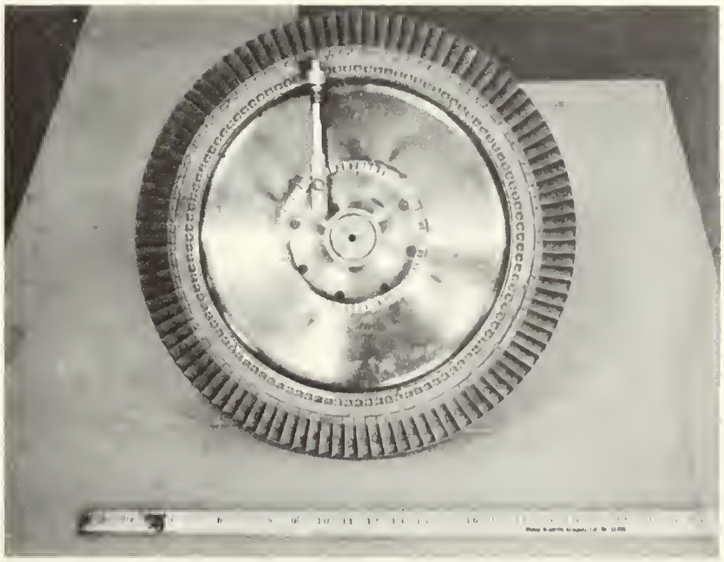
Bearing Housing
Figure II-1

Bearing Housing With Outer Races
Figure II-2



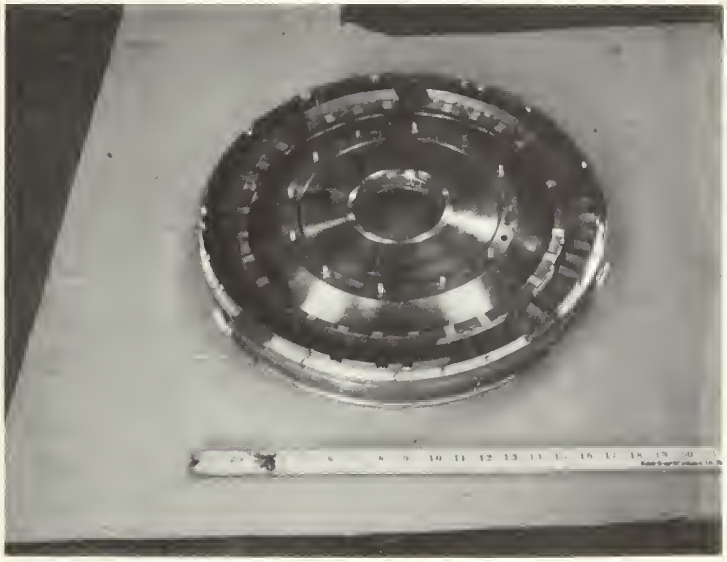
Upstream View Of Rotor

Figure II-3

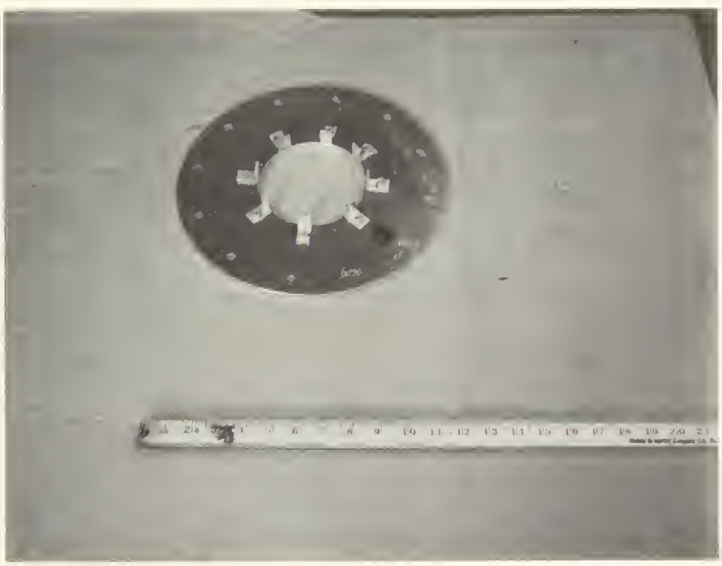


Downstream View Of Rotor

Figure II-4



Inlet Guide Vanes And Support
Figure 11-5



Alignment Plate
Figure 11-6

was pressed on to the rotor. During the final assembly, only the spline sleeve was pressed on to the spline shaft. A locking nut was placed on the end of the rotor shaft thus preventing any relative movement between the rotor and the bearing housing.

The location of the inlet guide vanes with respect to the rotor was maintained with a circular disk mounted on the end of bearing housing. Over-sized holes were drilled into the disk and the support brackets thus establishing radial and axial flexibility. Final alignment was achieved by utilizing feeler gauges.

Concurrent with rotor / stator assembly the test section was prepared for acceptance of the rotor. A circle of 16 3/8-inch holes and four 1.0-inch strut holes were drilled in the test section. Then, the rotor / stator assembly was inserted into the test section and aligned with the axis of the test section with the bolts supporting the guide vanes. The four struts were attached to the bearing housing and subsequently bolted to the shell.

Once the rotor / stator assembly was firmly mounted into the test section the dye and grease systems were installed. The base of the blade that was chosen to carry the dye was sealed since normal cooling air is not pumped to each blade separately. The rotor dye system proceeded from the discharge of the pump to the blade by way of a support strut and the hollow core of the rotor. The grease system tubing was inserted in one of the other struts. The inlet-guide-vane dye system was installed subsequent to the construction of the upstream contraction.

The foundation of the upstream contraction was formed from laminated pieces of styrofoam. A body of revolution was constructed with a third-order curve for a profile.

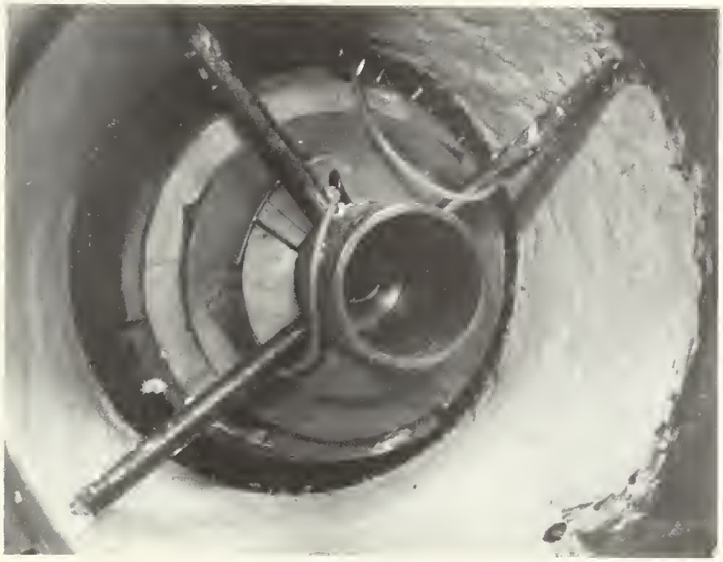


Figure II-7

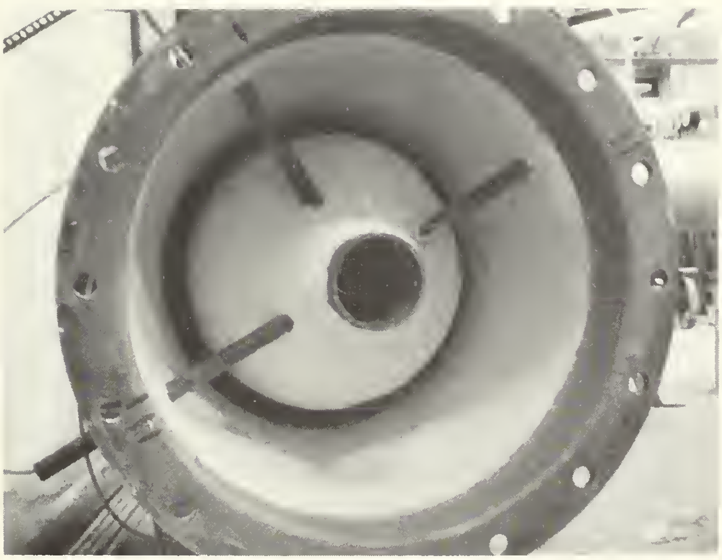


Figure II-8

The annular outer section was initially built with wood for reinforcement. Eventually the wood was removed and the cavities were filled with pieces of styrofoam (see results). The styrofoam surfaces were coated with plaster of Paris and in turn the plaster was sealed with a primer and marine epoxy paint. This completed the construction phase of the

experiment.

All auxiliary systems were tested prior to lifting the test section into the tunnel. The spray patterns and flow rates of the dye system were verified, and the grease system was pressurized. After a complete set of satisfactory tests, the test section was mounted into the water tunnel.

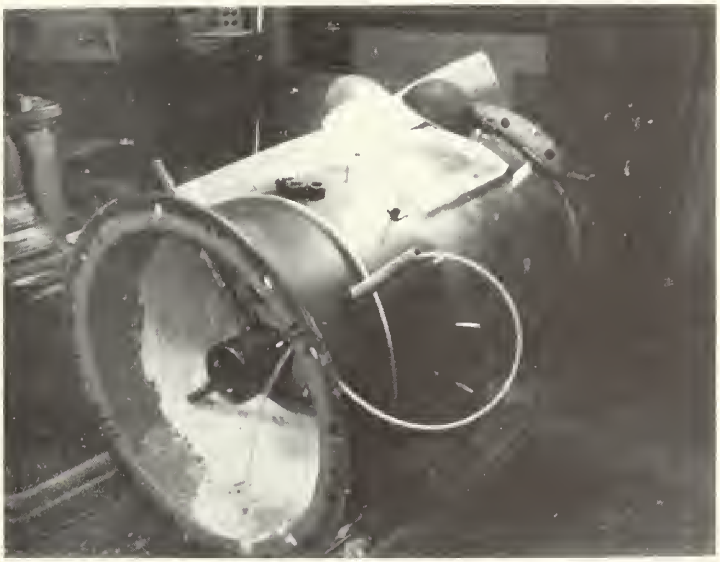
Several preliminary tests were conducted. During the original test runs, gasket leaks were repaired, back pressure was varied to determine realistic flow rates and turbine RPM for photographing the dye traces, and lighting schemes were investigated. An intermediate series of tests was conducted to ascertain the dye concentration most conducive to visualization.

The final tests were performed exclusively on the blade-tip cooling system. Dye flow rate and rotor speed was varied to determine their influence on cooling flow dispersion. The dye traces were photographed directly through the viewing part of the test section.

III. RESULTS

The test section with the gas turbine installed is illustrated in figures III- 1,2, and 3. The results of the first series of tests are illustrated in figures III-4,5, and 6. The final series of tests which resulted in photographs of the dye traces produced by the simulated coolant flow egressing from the rotor blade tips are illustrated in figures III- 7,8,9, and 10.

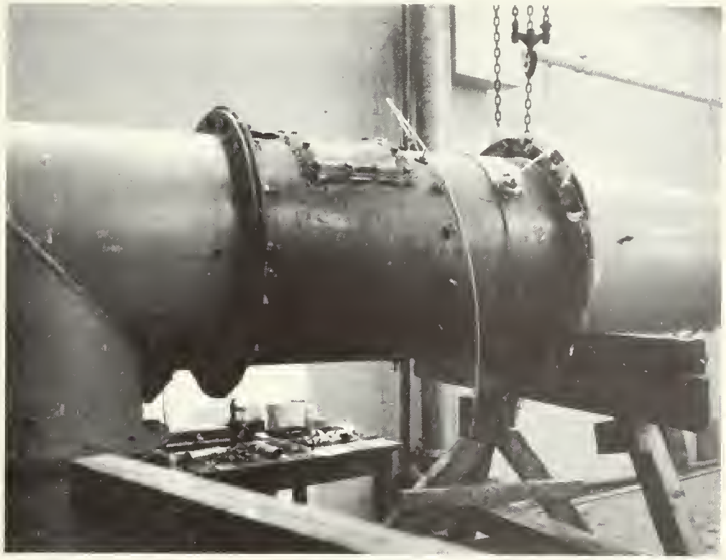
Figure III-1 shows the completed test section. The rotor and stator are in place, the acceleration surfaces are installed, the plexiglass viewing port is secured, and the struts containing the dye and grease systems are sealed.



Test Section
Figure III-1

Figure III-2 shows the test section installed in the water tunnel. A filter screen was mounted between the diffuser and the test section. All photographs of the

operating rotor and the dye traces were taken through the viewing port as shown in the figure.



Operating Configuration

Figure III-2

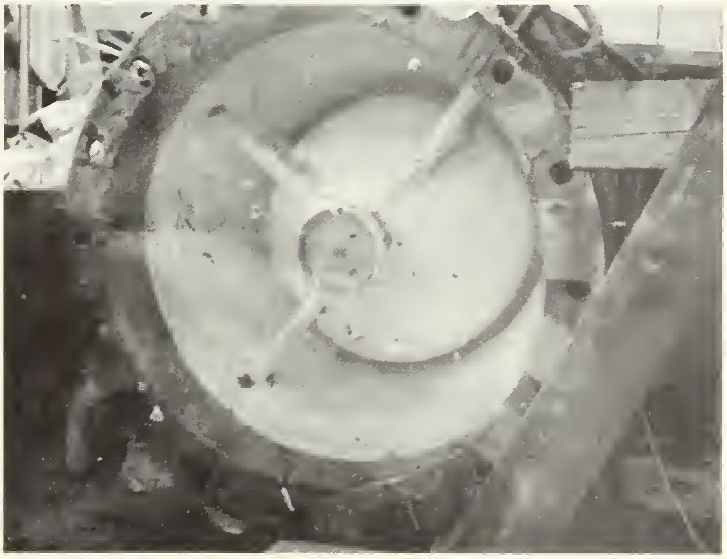
Figure III-3 shows the cavity down-stream of the rotor where the dye traces were photographed. The rotor is not

Downstream Cavity

Figure III-3

operating and the cavity is empty. This is the orientation of the camera and the light intensity employed during initial testing runs.

Figure III-4,5,6 show the test section after the first series of testing runs. Figure III-4 shows the test section just after it was removed from the water tunnel. The filter screen is still in place and the cracks in the plaster are barely distinguishable.



After First Tests
Figure III-4

Figures III-5 and 6 illustrate better the extent of the cracks in the contraction surface. No photographs of the damage to the downstream portion of the plaster were taken.

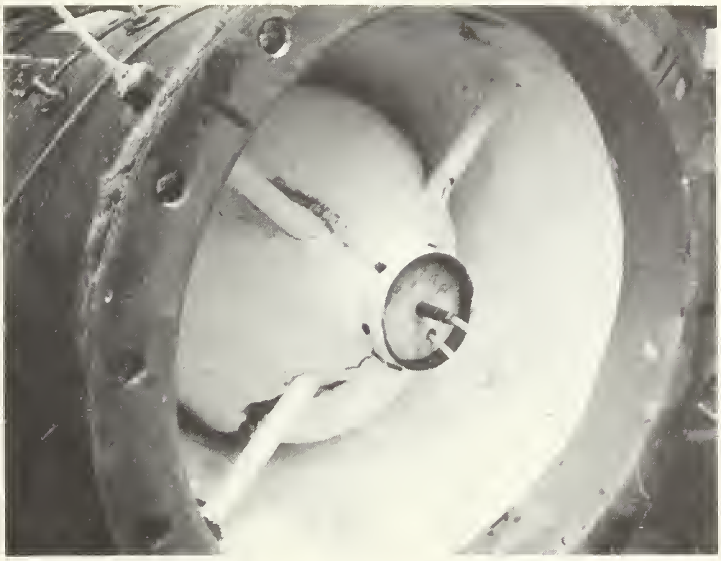


Figure III-5

Figure III-6

After the compressor surfaces were repaired, the test section was re-installed in the water tunnel and several series of tests were performed on the rotor. Figures III-7,8,9, and

10 are the results of these tests.

Dye Traces Run I
Figure III-7

Dye Traces Run II
Figure III-8

Dye Traces Run III
Figure III-9

Dye Traces Run IV
Figure III-10

Several additional photographs were taken; however dye visibility was very poor and the results were not included in this report.

IV DISCUSSION OF RESULTS

Several setbacks were experienced during the construction and the installation of the test section. The press-fittings of the different components of the support assembly for the rotor proved to be very difficult. Several of the surfaces that were pressed had to be re-machined. Interference fits above 0.001 to 0.0015 of an inch distorted the bearing races and caused malfunctions of the roller bearings. Additionally, the rotary O-rings produced a significant amount of torque. After the parts were re-machined, the torque produced by the support assembly was below the anticipated torque developed during the tests.

The bearing housing was machined to the O.D. of the bearings for only half its length. This simplified the machining; however it significantly complicated assembly. Eventually the bearing housing was bored out to the O.D. of the bearings for its entire length.

As outlined in the description of the apparatus, radial and axial alignment were controlled with the support plate mounted on the bearing housing. Considerable difficulty was experienced in achieving properly alignment without an occasional rub between the stator and the rotor. It was evident at this point that a more sophisticated adjustment system would have eliminated the many hours spent on alignment.

The support of the inlet guide vanes with a radial circle of bolts proved to be far superior to what was anticipated. The bolts formed such a rigid structure that the four large struts at the upstream end of the bearing housing were not necessary. The bolts did have a tendency to cant the individual guide vanes if the bolt was not positioned close to the center of the vane. This added to the problem of maintaining proper blade-tip clearances.

The dye systems functioned well during the tests; however the isolation of individual blades was rather complicated. Normally, the coolant enters from large plenums: thus the hollow blades are not air or water tight.

The original contraction surface was built with a wood foundation. It was anticipated that the plaster and the epoxy sealer would prevent the water from reaching the wood and causing any distortion. Figures III-5 and 6 indicate the damage resulting from the wood swelling. The seizing of the rotor was caused by the wood in the outer annulus expanding and compressing the inlet guide vanes onto the rotor. All wood was subsequently removed and replaced by styrofoam. Styrofoam proved to be a very rigid, watertight foundation for the acceleration surfaces. No problems were experienced with the styrofoam during the second series of tests. Some of the plaster used to fair in struts and dye tubes did break off during the final tests. This, nowever, did not jeopardize the tests.

The photographic results achieved during the final tests were of marginal benefit and quality. Lighting, refraction, and parallax vitiated the resolution and definition of the photographs. Although the dye traces were visible to the eye, the camera was incapable of reproducting them on film. The viewing port limited the amount of light that could illuminate the field of view. Consequently, there were conflicting photographic requirements. On the one hand, reduced exposure time was desirable due to the movement of the rotor, and on the other increased exposure time would have admitted more light to the film. Lighting and camera limitations thus prevented the achievement of satisfactory photographs.

The visualization of secondary-flow phenomena was dependent on coolant (dye) flow rate, turbine speed, and

flow rate. The dye patterns observed cannot be considered indicative of normal turbine operating conditions. Low-speed flow conditions suitable for flow visualization were considerably below those for Reynolds similarity. Furthermore, the abrupt expansion of the flow after it passed through the rotor produced unrealistic flow patterns.

flow rate. The dye patterns showed a series of [illegible]

measurements indicative of [illegible]

conditions. Increased flow [illegible]

visualization was [illegible]

similarity. Furthermore, the [illegible]

after it passed through the [illegible]

flow patterns.

V CONCLUSIONS

Although the apparatus designed and partly developed for the visualization of secondary-flow phenomena in cooled gas-turbine blades functioned well, the actual construction would have been simplified if emphasis was placed on maintenance and repair. Any trouble-shooting required an inordinate amount of time to perform. Additionally, special tools had to be built to assemble and disassemble the rotor assembly. The capability to repair or inspect any component should have been incorporated in the design.

As mentioned previously, rotor alignment and blade-tip clearance were extremely sensitive. A more positive means of control such as employed in the chuck of a lathe would have facilitated alignment. Positioning of the rotor assembly in the test section was highly dependent on the location of the struts and the radial bolts. This was accomplished manually with rather crude instruments. Mechanical layout and construction would have been far more desirable, i.e. the test section should have been drilled and tapped in a lathe.

The problems encountered with machined parts could have been precluded if assembly of the rotor housing was performed in the machine shop. This would have assisted in the pressing of the bearings and the O-ring supports.

The styrofoam foundation for the contraction surface proved to be far superior to the wood. If more time was available, it would have been highly desirable to contour the surface flush with the edge of the inlet guide vanes and thus eliminate the need for plaster. Due to the many interferences, the construction of such a surface was not compatible with the time schedule.

It is evident from figures B-7,8,9, and 10 that considerable work has to be done to improve the quality of the photographs. If lighting and optical problems were solved, it is still questionable whether the results would be meaningful. Only at extremely low flow rates would streamlines be visible. Under these circumstances considerably more laminar flow would be present than is normally found in the operating turbine. Thus the ultimate objective of the investigation, the duplication of momentum transfer in the blade passages, would not be attained. However further research could be of benefit to the theoreticians since current mathematic modes are predicated on very little empirical data.

VI. RECOMMENDATIONS

The open-channel water tunnel employed in this experiment possessed several valuable characteristics; however there were some features that warrant improvement. The plate valve on the end of the discharge pipe could not be operated while water was flowing through the tunnel. A more sophisticated turning mechanism with possibly a motor-operated hand wheel is recommended. The remotely-controlled gate valves in the pumping system were difficult to control. In order to cope with this deficiency, two persons were always present when the rig was operating. This is a major disadvantage and would require extensive modifications to correct.

Considerable difficulty was experienced in photographing the dye traces in the water. It is recommended that thought be given to installing a permanent lighting system in the tunnel downstream of the rotor. Space is available, and this would reserve the viewing port exclusively to photography. Additionally the inner surface of the test section should be painted a color more conducive to dye detection. Use of the coherent fiber-optic bundle was not compatible with the time schedule of this experiment. The use of this test alone could produce a significant improvement in the quality of the photographs.

As mentioned in the conclusion, it would be desirable to eliminate entirely the use of plaster of Paris. The styrofoam would have to be sealed with an alternative substance since the epoxy paint reacted with the styrofoam core.

In retrospect, the selection of material for application in conditions similar to those experienced in this experiment is very important. Frequently the most

expedient, economical choice is not the most desirable. In turn thought must be given to the dynamics of the situation. Seals and joints must be constructed properly, for nominal pressure gradients will cause leaks if the mating surfaces are not compatible. Brute-force, one-inch-thick gaskets, and gobs of sealer should be avoided if time and money permits.

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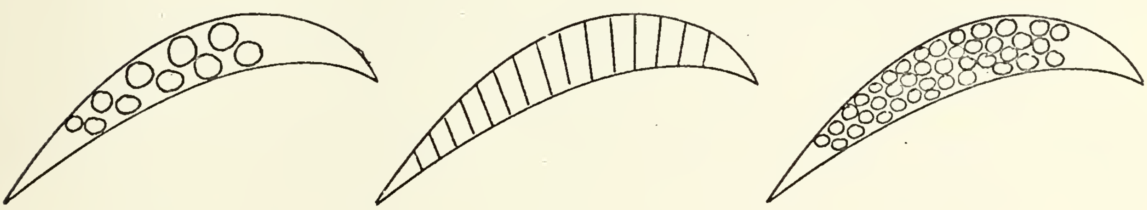
VIII. APPENDICES

A. SUPPLEMENTARY BACKGROUND INFORMATION

The National Advisory Committee for Aeronautics has sponsored many investigations relevant to the feasibility and the effectiveness of different blade-cooling techniques. Initial work dealt with analytical estimates of the benefits of increased turbine-inlet temperatures. Other independent endeavors by Brown [1] and Burke [7] have resulted in a common conclusion. If a cooling system can be adapted efficiently to a gas turbine, significant increases in power and reduction in size and fuel consumption could be achieved.

Once it was determined that cooled turbine blades represented a promising solution to the problem of higher inlet temperatures, several different cooling systems were developed and subsequently evaluated in experimental turbines.



(6) 10 Tubes | (8) 15 Fins | (9) 34 Tubes

Figure I

Figure I illustrates three of the first concepts developed. In each of the three cases, the blades were built from hollow shells with either tubes or fins installed to guide the flow of the cooling air and to enhance heat transfer. All three proposals were effective in lowering the temperature of the blade at the mid-chord point on the pressure and suction side of the blade; however none of the

schemes were capable of lowering the temperature at the leading or the trailing edges. It would appear that all of these cooling systems would discharge a substantial amount of air into an operating turbine thus increasing compressor demands and disrupting the desirable flow characteristics of the turbine. Ultimately excessive cooling air could vitiate the improvements derived from higher T.I.T.'s. Since only four blades, each of the type illustrated above, were installed in experimental turbines, the effect of the cooling system on overall turbine performance was not evaluated. Subsequent tests were performed by NACA on flat plates with transpiration, film, and convection-cooling systems [10]. In order to compare the effectiveness of each scheme, gas velocity and gas temperature were maintained constant; however laminar and turbulent flow with and without radiation were simulated. For laminar flow it was determined that three times as much coolant flow was required for optimum convection cooling as was required for transpiration cooling. The difference in coolant requirements increased disproportionately for increasing coolant flow. Hence in applications where strong cooling is required, transpiration cooling is far superior to convection cooling. If an abnormal amount of radiation is present, the advantages of transpiration cooling are reduced. In turbulent flow the same characteristics were observed; however at optimum convection cooling the quantity of air required was only 1½ times the quantity for transpiration cooling. The key feature of film cooling was that it could effectively cool specific locations and could approach the effectiveness of the other methods by increasing the number of slots. [10]

Concurrent with the investigation of the cooling system was a program at NACA to determine the nature of secondary flows in turbomachinery [11,12]. The objective of this research was the verification of current predictions concerning the nature of secondary flows. Squire and Winter [13] and

Hawthorne [14] derived expressions for the value of the secondary component of vorticity. Experiments on cascades conducted by Armstrong [15], Horlock [16], and Louis [17], have substantiated this theoretical work if there are small viscous effects and no separtion. A photographic study by NACA on a series of blades in a cascade producted some very valuable results[11,12]. By passing air through the cascade and varying the position of a smoke probe, several photographs at different orientations were taken. The photographs verified the existence of a corner vortex located in the corner formed by the suction surface of the blade and the cascade wall. The pattern appeared to originate well up in the passage and was definitely not a trailing-edge phenomenon. The researchers questioned the applicability of current two- and three-dimensional analyses of flow patterns in the wall region due to the complexity of the flows observed, and they concluded that secondary flow had a profound effect on the principal flow patterns in the cascade.

More recently, Senoo, Yamaguchi, and Nishi [18] attempted to photograph three-dimensional flows in a radial compressor. The compressor was operated at very low Reynolds numbers in order to facilitate the utilization of flow-visualization techniques. The photographs obtained were extremely effective in illustrating secondary-flow phenomena. The superposition of a series of photographs where the dye was admitted at different positions was particularly informative since bulk-flow-pattern interactions were very evident. The authors of the papers felt that even though Reynolds similarity was not maintained during the test the results were indicative of normal operating conditions. The authors further concluded that secondary flow varies with operating conditions and the clearance between the casing and the impeller, and this complicated

phenomenon can be synthesized from a few, well-known simple secondary flows e.g. corner vortex.

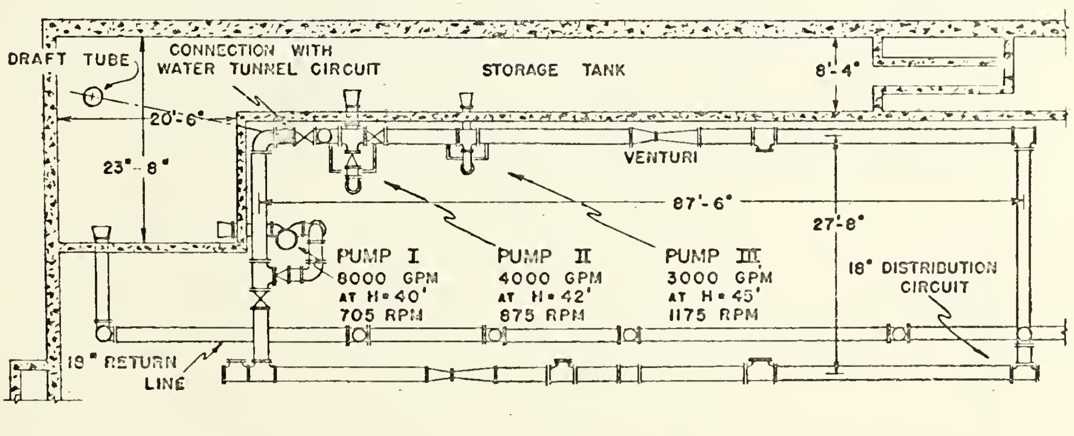
Several unanswered questions still require investigation. Even though secondary flows were visualized in an axial-flow turbine cascade, the influence of centrifugal force was not prevalent. This force modified the value of the local heat-transfer coefficient when results for an operating turbine [2] were compared with a stationary cascade [3]. Thus it is not unreasonable to expect a similar deviation from the results obtained by Herzig [11]. Futhermore the influence of inlet nozzles and a second-stage stator have not been included in any of the previous investigations. Senoo [18] observed that the development of secondary flows was strongly dependent on the clearance between the casing and the impeller. Thus the presence of the nozzles and stator could significantly modify earlier results. Finally, the contribution made by the cooling flow to the developing flow patterns of the turbine have heretofore not been photographed.

B. DESCRIPTION OF APPARATUS

The open-circuit water tunnel located in the Hydrodynamics Laboratory was the testing facility that was utilized for the present experiment. The advantages given by this system were the following: adequate flow rate, geometric compatibility, and exclusive access for the duration of the testing period.

The open-circuit water tunnel was originally designed with a 7½ inches by 9 inches closed-jet working section with a maximum velocity of approximately 50 fps [19]. There are two independent flow systems which supply the water tunnel. The systems receive water from three centrifugal pumps which have rated capacities of 8,000; 4,000; and 3,000 gallons per minute at heads of 40 feet. The flow rate is remotely controlled by means of motor-operated values located on the discharge side of the pump. The three pumps take suction from a 75,000 gallon storage. Figures B-I, B-2, and B-3 indicate the general arrangement of the system.

The water tunnel as viewed in figure B-I was modified to accommodate the gas turbine (maximum O.D. 21.0 inches). The standard test section was removed, holes were drilled in



Pumping System

Figure B-1 (19)

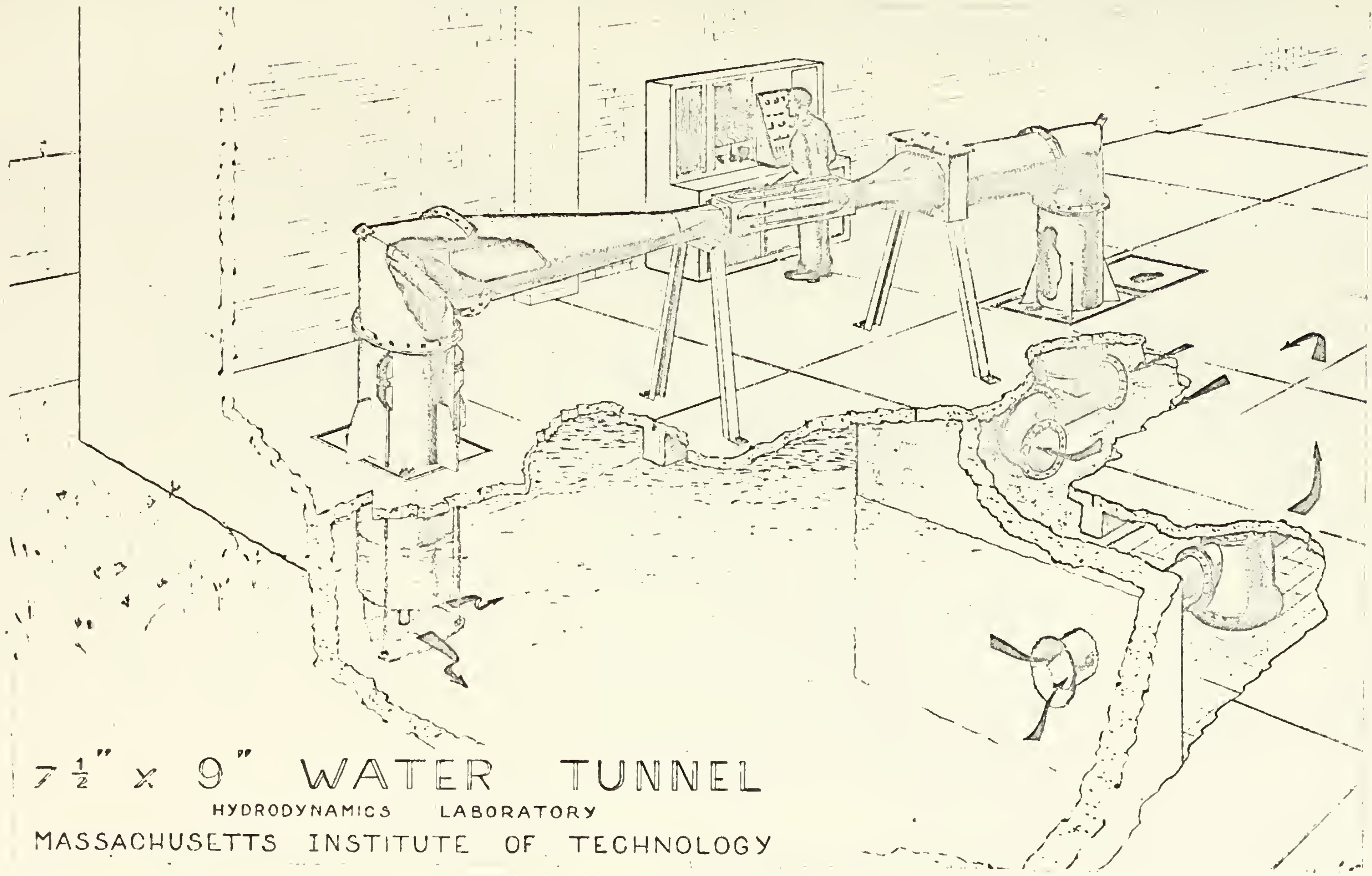

7 1/2" x 9" WATER TUNNEL
HYDRODYNAMICS LABORATORY
MASSACHUSETTS INSTITUTE OF TECHNOLOGY

Figure B-2 (19)

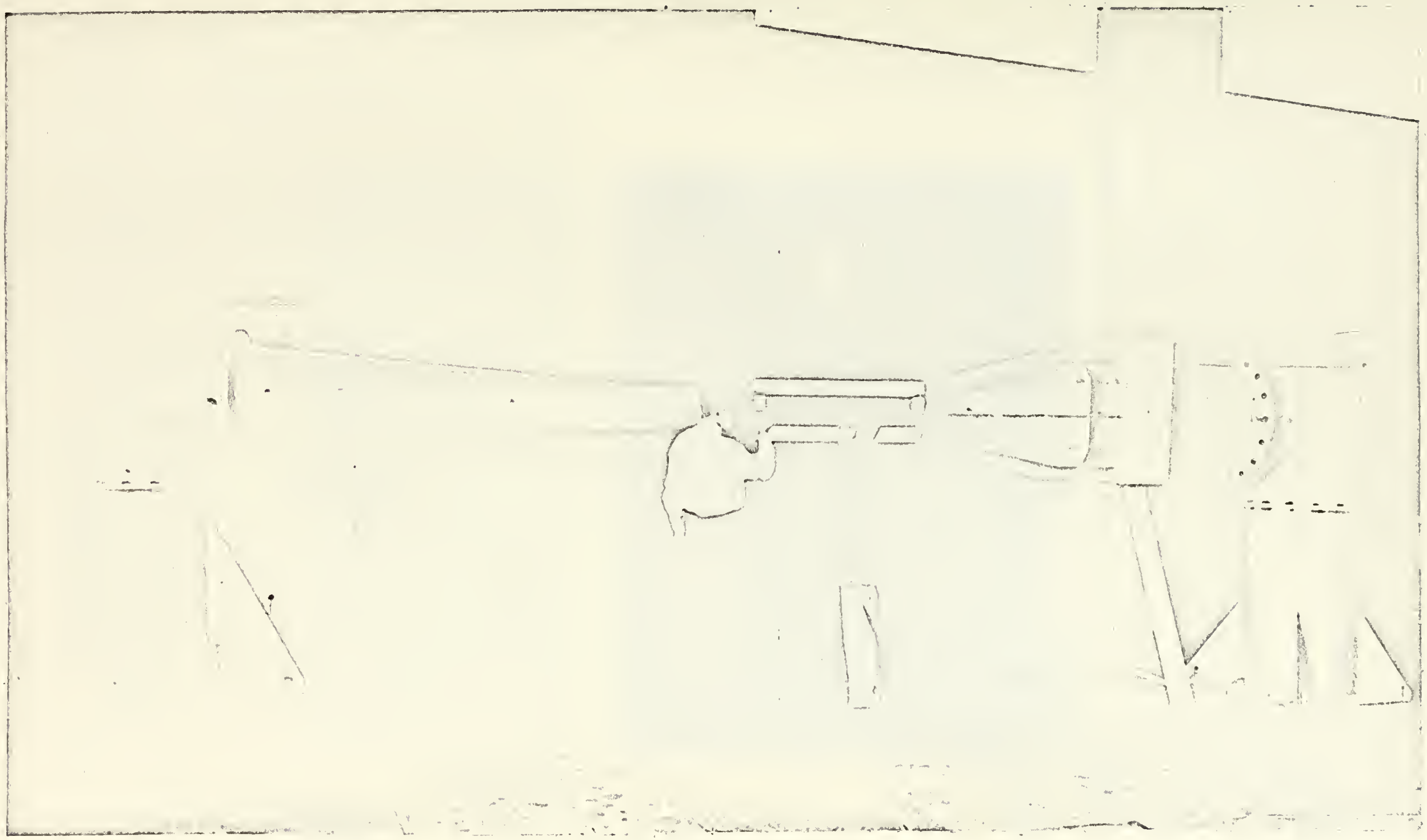


Figure B-3 (19)

the flanges of the adjacent sections to provide for securing bolts, and the 9.0 foot diffuser was joined with the contraction section. The original downstream support frame was re-located and an additional vertical support was installed to augment the supporting capacity of the tunnel. The new arrangement increased the working diameter of the test section from approximately 8.0 inches to 23.5 inches, sufficient space for the turbine.

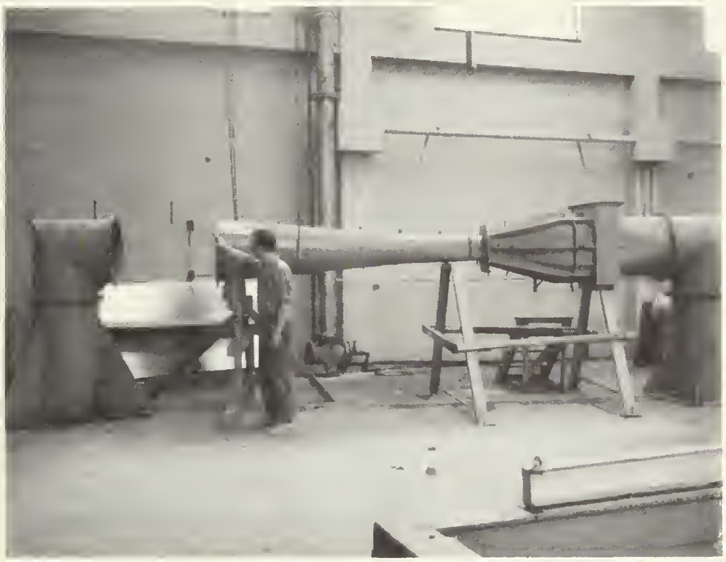


Figure B-4

Several components of the basic tunnel proved to be very beneficial during the tests. The four filter screens located just upstream of the contraction removed foreign matter from the flow, attenuated eddies created by the 24-inch elbow, and smoothed the velocity distribution. The guide vanes installed in each of the 90° elbows minimized separation and suppressed secondary flows. The permanently installed venturi (see figure B-3) provided a means of measuring flow rate. Finally, the plate valve mounted on the discharge pipe regulated back pressure in the tunnel and thus prevented the inception of cavitation. Furthermore, the plate established an even, radial discharge of the flow into the resevoir which in turn

provided a steady suction head for the circulating pumps.

The new test section made for the present experiment was fabricated from 3/16- inch rolled-steel plate. The thickness of the plate was in accordance with original tunnel design criteria. Dimensions of the section were dicated by the space remaining after modification of the tunnel and the size of the flange on the diffuser. A 9.0 inch by 12.0 inch viewing port was included in the test section to facilitate minor repairs and to monitor overall turbine operation. The longitudinal joint of the test section was welded on both sides while the 3/8 inch flanges were welded completely on the inside and tack welded only on the outside. This was in keeping with tunnel-design procedures. Non-destructive tests were not performed on the welds.

Figure B-5

A ¾-inch plexiglass plate was manufactured to cover the viewing port. The plate was heat treated and then bent

to the desired curvature. The curvature was subsequently modified during installation with heating lamps since the plate apparently changed curvature when cooling.

The Allison 501 gas turbine with first-stage cooling was the turbine employed in the tests. The inlet guide vanes and the first-stage rotor are built with cooling cavities to which compressor bleed-off air is supplied.

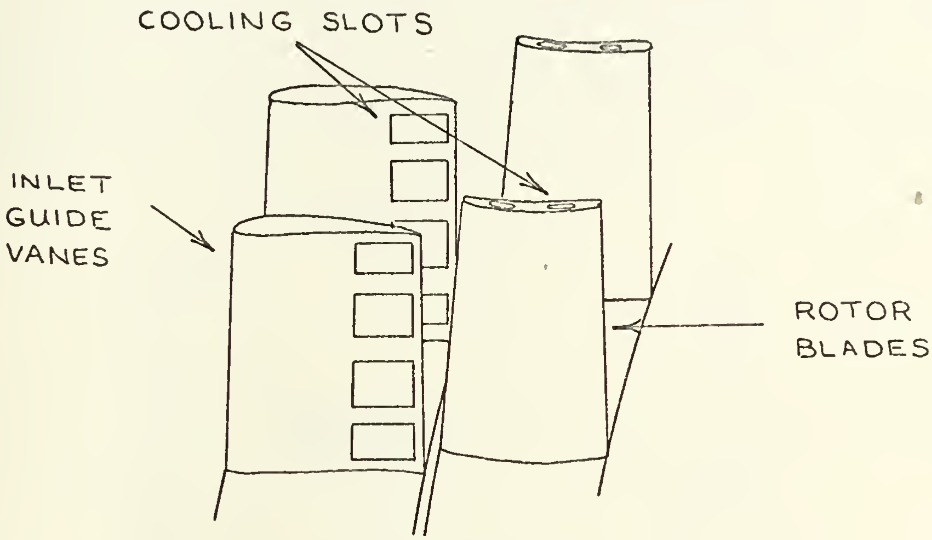
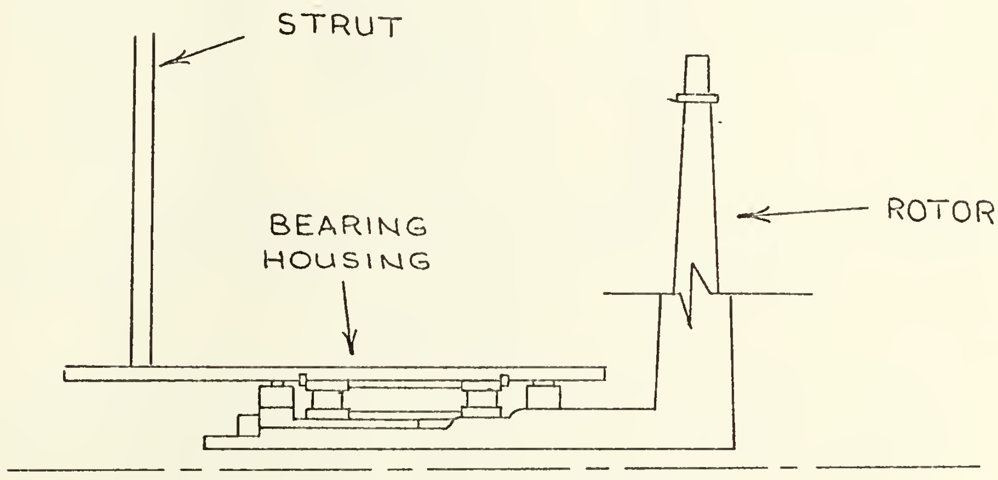


Figure B-6

The inlet guide vanes, guide-vane supports, and first-stage rotor were mounted in the test section with primary emphasis on duplicating normal operating conditions. The rotor was supported by a pair of tapered-roller bearing that were installed in a fabricated bearing-housing sleeve. The down-stream bearing was located on the rotor's normal bearing surface while the upstream bearing was pressed on to a brass sleeve. The brass sleeve was in turn pressed on to the spline shaft of the rotor. Two brass spacers were installed between the bearings to maintain the proper relative positions of the inner and the outer races of the bearings under rotor thrust. The bearings were rated at 3,500 ft-lb, well within the

limits of the tests (see Appendix D.). To preserve the bearings, the entire bearing assembly was enclosed in a pressurized grease chamber. The grease was contained by the rotor, the bearing housing, and two brass spacers. The spacers acted as supports for two rotary o-rings. The o-rings were rated at 800 psi pressure differential and 500 rpm. The system was pressurized remotely through poly-flo tubing with a standard grease gun. Rotor thrust was transmitted to the test section through a locking nut, the bearings, the retainer rings, the bearing housing, and finally the support struts.



Rotor Support Assembly

Figure B-7

The inlet guide vanes and guide-vane support were supported by a plate mounted on the end of the bearing housing. The oversize holes in the bolt circle of the support plate were for adjusting the radial position of the guide vanes with respect to the rotor. The series of brackets that attached the support plate to the bearing housing had a slot in each for adjustment of axial alignment. The guide vanes were held in position by three steel strips around the periphery of the assembly.

Additionally, the inlet guide vanes were supported by a series of radially oriented bolts. These bolts provided not only vertical support, but also furnished support against the thrust developed by the rotor and stator and assisted in aligning the assembly with the axis of the test section.

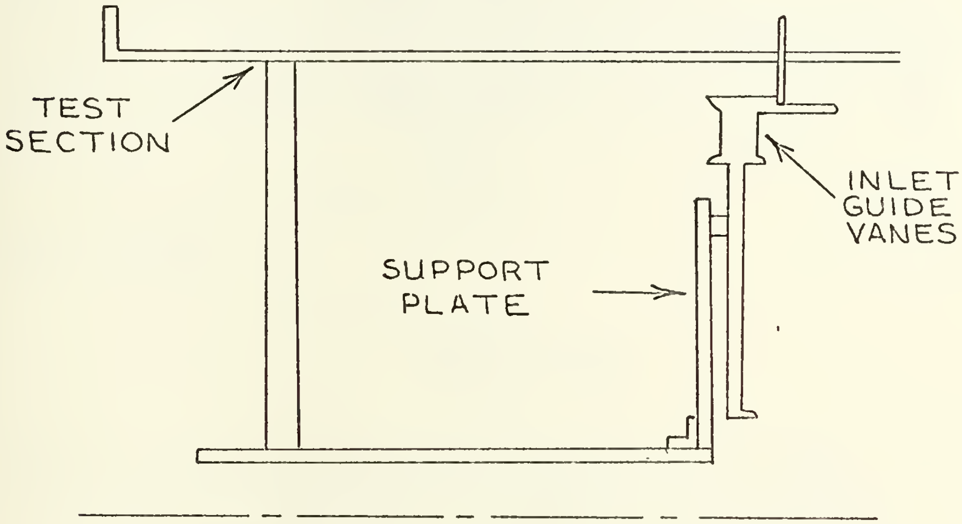
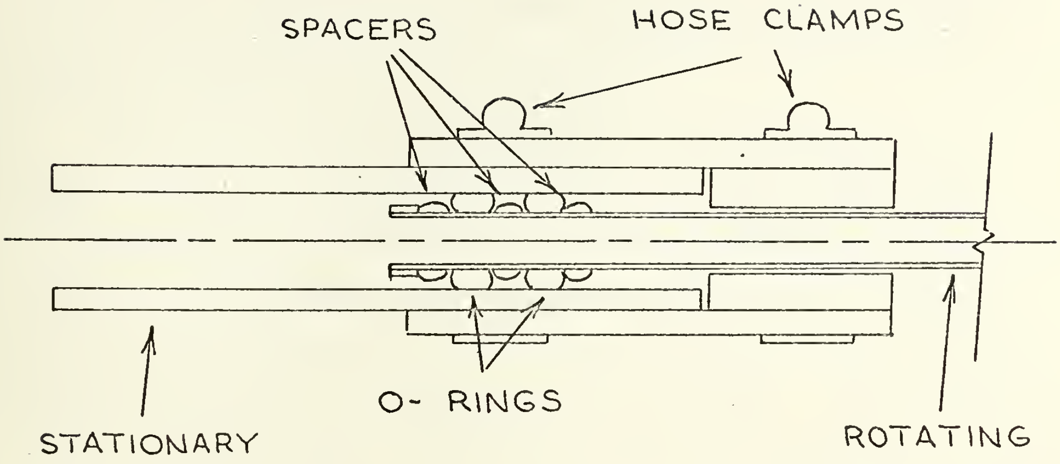


Figure B-8

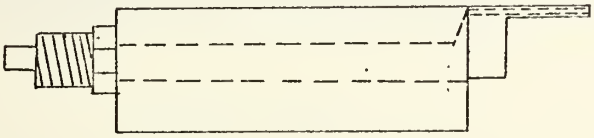
The three struts on the upstream end of the bearing housing provided vertical support for the bearing housing and functioned as conduits for the poly-flo tubing that carried the dye and the grease. Two independent dye systems were installed to carry dye to the inlet guide vanes and to the first-stage rotor. Each dye system required special fittings since normal cooling air enters the blades from a large plenum which could not be used in the present arrangement. On the rotor, the test blade was sealed and a small hollow needle injected the dye into the cavity. The needle was adapted to a standard 90° elbow fitting. The dye was pumped from a small reservoir to the fitting through a series of pieces of poly-flo tubing. A special rotary fitting with o-ring was built to transfer dye

to the rotor.



Rotary Seal
Figure B-9

Construction of the dye system for the inlet guide vanes was complicated due to the narrow opening in the base of the blade that admits the cooling air. A special adapter was built that inserted into the guide-vane orifice on one end and accepted a standard poly-flo fitting on the other. A single piece of tubing carried the dye directly from the discharge of the pump to the guide-vane adapter. The tubing was imbedded into the surface of the upstream flow contraction.



Inlet Guide-Vane Dye Injector
Figure B-10

Each dye system was serviced by a separate pump, and the pumps took suction through a filter system from a single reservoir. Potassium permanganate was the dye employed in the tests because it eventually bleached out and thus did not contaminate the reservoir.

The flow accelerated smoothly into the inlet guide vanes through a contraction upstream of the guide vanes. The inner hub was mounted on the bearing-housing sleeve and the outer surface was situated between the inner wall of the test section and the tips of the inlet guide vanes. The foundation of the surface was styrofoam. The styrofoam core was sealed with plaster of Paris and the plaster surface was coated with marine epoxy paint. The epoxy paint was impervious to water and exhibited excellent bonding strength.

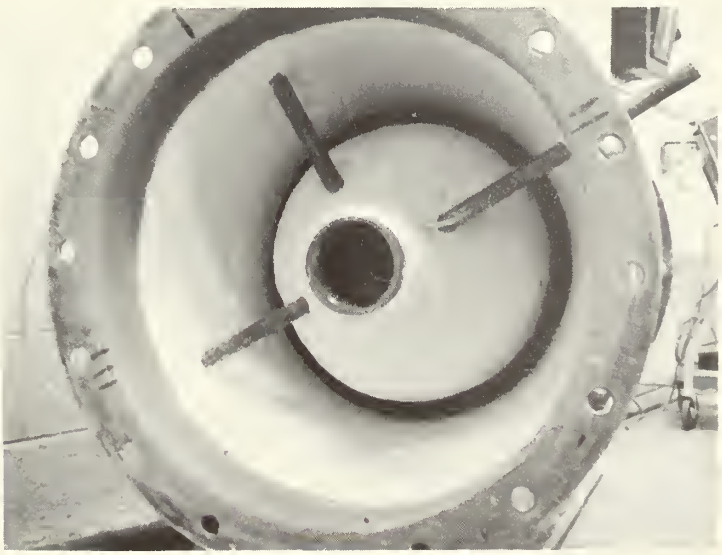


Figure B-11

All photographs were taken with a Poloroid 350 camera. A special close-up attachment was utilized in the photographing of the dye traces.

C. SUMMARY OF RESULTS

The following is a chronological listing of the salient features of each test run and a brief statement of the results:

Test	Date	Duration	Results
1	5 May	3 Min.	Large leak developed between contraction section and diffuser, secured to accomplish repairs
2	7 May	10 Min.	Unable to fill cavity downstream of rotor, several heavy leaks around view port and strut holes, noted rather severe rub in turbine, secured to adjust back-pressure plate and repair leaks
3	7 May	1.5 Min.	Rubbing intensified, rotor seized, secured to investigate problem
4	11 May	8 Min.	Several heavy leaks existed around viewing port, support struts, and upstream test section flange; secured to repair
5	12 May	15 Min.	Some leaks still existed around viewing port and support strut; back pressure too low; secured to correct
6	12 May	15 Min.	Leaks still present however tolerable, back pressure and flow rate improved, rotor turning smoothly, no rub, secured to set up dye systems.

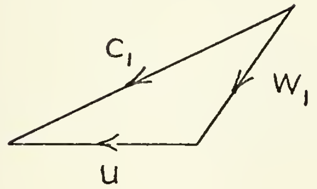
Test	Date	Duration	Results
7	12 May	30 Min.	Dye systems tested, dye traces were very weak; however systems functioned well, back-pressure plate needed adjustment, leaks tolerable, secured to darken dye and adjust plate
8	12 May	30 Min.	Dye traces still very weak, parallax prohibited viewing of inlet-guide-vane dye traces, back pressure satisfactory, lighting schemes were evaluated, secured to darken dye and improve lighting
9	12 May	22 Min.	Varied dye flow rate and rotor RPM, dye concentration insufficient to photograph, contrast very poor, secured to adjust dye system
10	12 May	20 Min.	Apparently sediment from the bottom of reservoir was entrained in the flow; water was very cloudy; secured to let reservoir settle.
11	13 May	40 Min.	Installed new lighting system, new system improved dye trace visibility, took several photographs with close-up lens on camera, varied lighting and exposure time, unable to pick up dye traces in photographs.
12	13 May	60 Min.	Supplemented lighting system with addition light, employed standard

Test	Date	Duration	Results
			flash attachment, resolution and definition still very poor, dye traces clearly visible, secured to improve lighting
13	13 May	75 Min.	Photographed dye traces, definition barely satisfactory, lighting still needs improvement, parallax prevented photographing of dye egressing from inlet guide vanes, secured to consider new lighting systems

Note: when the tunnel was secured, the pumps were shutdown and all liquid was drained from the system.

D. Summary Of Calculations

Design-point operating characteristics for the Allison 501 gas turbine:

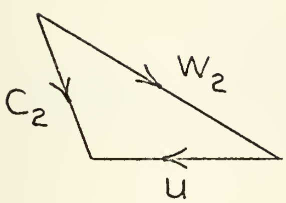


$C_1 = 1631$ fps, $M = 0.769$

$W_1 = 878$ fps, $M = 0.414$

$C_{x_1} = 718$ fps,

Inlet Velocity Triangle
Figure D-1



$C_2 = 1464$ fps, $M = 0.708$

$W_2 = 802$ fps, $M = 0.388$

$C_{x_2} = 743$ fps

Outlet Velocity Triangle
Figure D-2

u = 960 fps

nozzle tip dia. = 17.0 in.

nozzle hub dia. = 14.8 in.

N = 13,820 rpm

inlet area = 0.378 sq. ft.

mass flow rate = 39.0 lbm / sec.

blade chord = 0.626 in. = 0.0521 ft.

Determination of operating conditions at Reynolds similarity:

assuming for the combustion gases entering the rotor K = 1.33,

$$a = \sqrt{kRg_cT}$$

and

$$M = \frac{V}{a}$$

therefore $T = \dfrac{V^2}{k\ R g_c\ M^2}$

or T (static, rotor inlet) = 1975 $^\circ$ R

similarly for rotor outlet

$$T = \frac{V^2}{k\ R g_c\ M^2}$$

or T (static, rotor outlet) = 1880 $^\circ$ R.

Given, $\dot{m}$ = 39.0 lbm / sec but $\dot{m} = \rho V A$

$$= \frac{\dot{m}}{VA} = \frac{39.0}{(718)(0.378)}$$

(stator inlet) = 0.1438 lbm / cu ft.

$$\left(\frac{\rho_1}{\rho_2}\right)^{\gamma-1} = \frac{T_1}{T_2} = \frac{1975}{1880} = 1.05$$

therefore (rotor outlet) = 0.1240 lbm / cu. ft.

Computing Reynolds number for normal operation,

$$Re_{norm} = V L \rho / \mu$$

$$Re_{norm} = \frac{(1464)\ (0.1240)\ (0.0522)}{(2.85 \times 10^{-5})}$$

$$Re_{norm} = 3.32 \times 10^5$$

In order to maintain Reynolds similarity,

$$Re_{norm} = Re_{exp}$$

therefore $Re_{exp.} = 3.32 \times 10^5$

or $V = \dfrac{Re_{exp}\ \mu}{L \times \rho}$

V = 66.6 fps.

The inlet axial velocity at Reynolds similarity,

V = 32.7 fps

assuming (experimental) = 62.4 lbm / cu ft.

therefore $\dot{m} = \rho V A$

$\dot{m}$ = (62.4) (32.7)(0.378)

$\dot{m}$ = 744 lbm / sec.

or $\dot{m}$ = 5,770 gpm.

Determination of torque developed at Reynolds similarity:

from Euler's Equation

$$T \text{ (net torque)} = \frac{\dot{m}}{g_c}\left(r_1 C_{\Theta_1} - r_2 C_{\Theta_2}\right)$$

$\dot{m}$ = 5,770 gpm

r ave = 7.95

c_{Θ_1} = 66.8 fps

c_{Θ_2} = 13.8 fps

Therefore Torque $= \frac{47,900}{32.2} \times \frac{7.95}{12.0} \times (66.8 - 13.8)$

Torque = 870.0 ft - lbs.

Calculation of rpm's from inlet velocity triangle,

$$\frac{U \text{ exp}}{960} = \frac{32.7}{718}$$

or $\quad$ U exp = 43.7 fps

$$N = \frac{U \times 60}{d}$$

$$N = \frac{43.7 \times 60}{3.14 \times \frac{15.9}{12}}$$

N = 630.0 rpm.

Results for Reynolds similarity:

U (inlet axial velocity) = 32.7 fps

$\dot{m}$ = 5,770 gpm = 744 lbm / sec.

N = 630.0 rpm

T (net) 870.0 ft - lbs

Variation of operating conditions with inlet axial velocity:

$$\dot{m} = \rho V A$$

since ρ and A are constant then

$$m = f(V) = K_1 x V = K_1 x C_x$$

where $\quad K_1 = xA.$

Variation of torque :

$$T = \frac{\dot{m}}{g_c}\left(r_1 C_{\Theta_1} - r_2 C_{\Theta_2}\right)$$

assuming inlet and outlet velocity triangles remain similar at reduced speed then C_{Θ_1} and C_{Θ_2} are functions of c_x (inlet axial velocity),

therefore $T = \rho \frac{VA}{gc} xr \; x \Delta C_{\ominus}$

but $\Delta C_{\ominus} = k_2 \; c_x$

$$T = K_3 \; C_x^{\;2}$$

where $K_3 = \frac{A}{gc} xrx\Delta C$.

Finally, $N = \frac{U \times 60}{d}$

and $N = K_4 \; c_x$

where $K4 = \frac{60}{d} \times K_5$ and K_5 is determined from the inlet velocity triangle.

$$K_1 = \rho A = (\ 62.4\ )\ (\ 0.378\ )$$

$$K_1 = 23.6$$

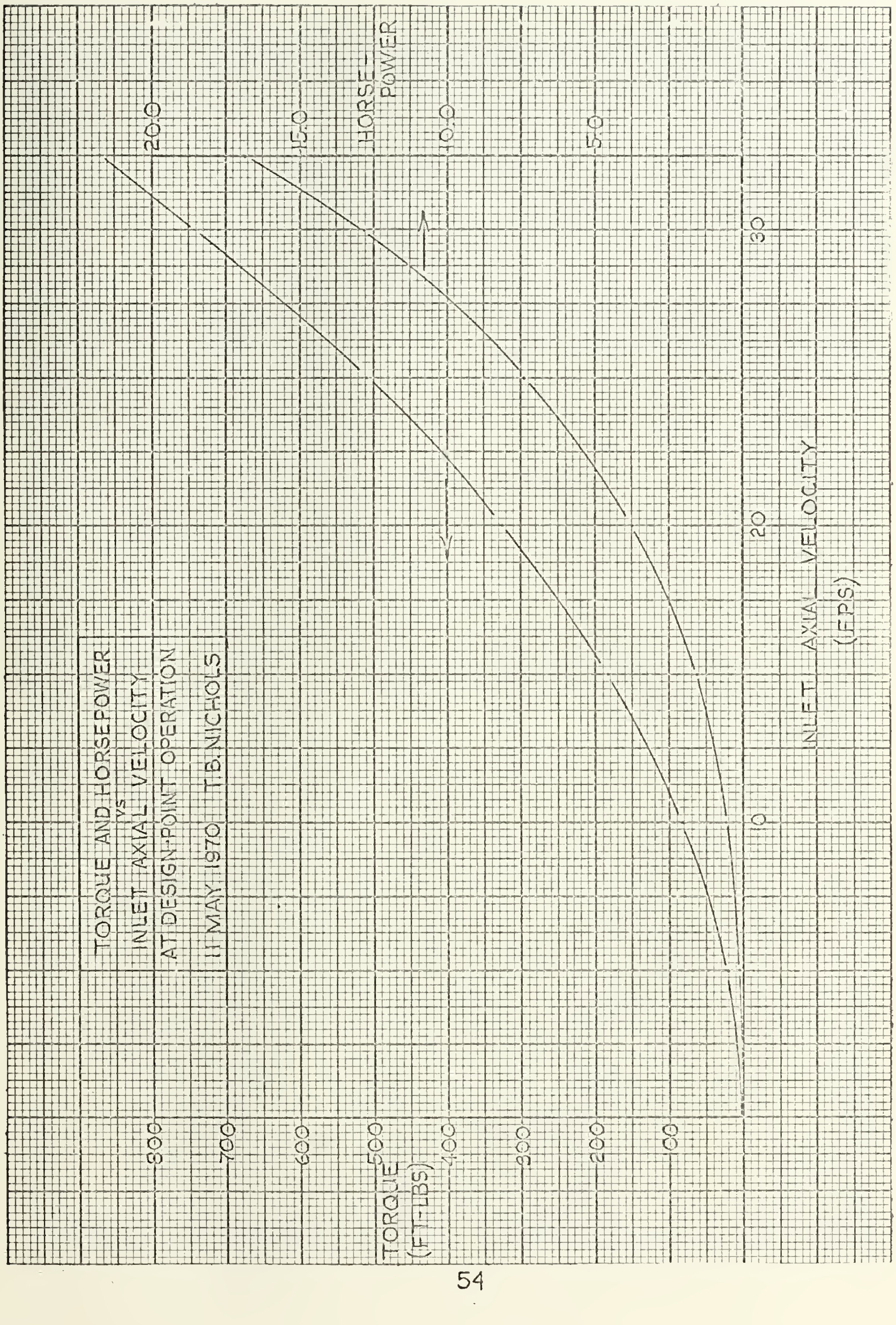
$$K_3 = \frac{T}{C_x^2} = \frac{870}{(32.7)^2}$$

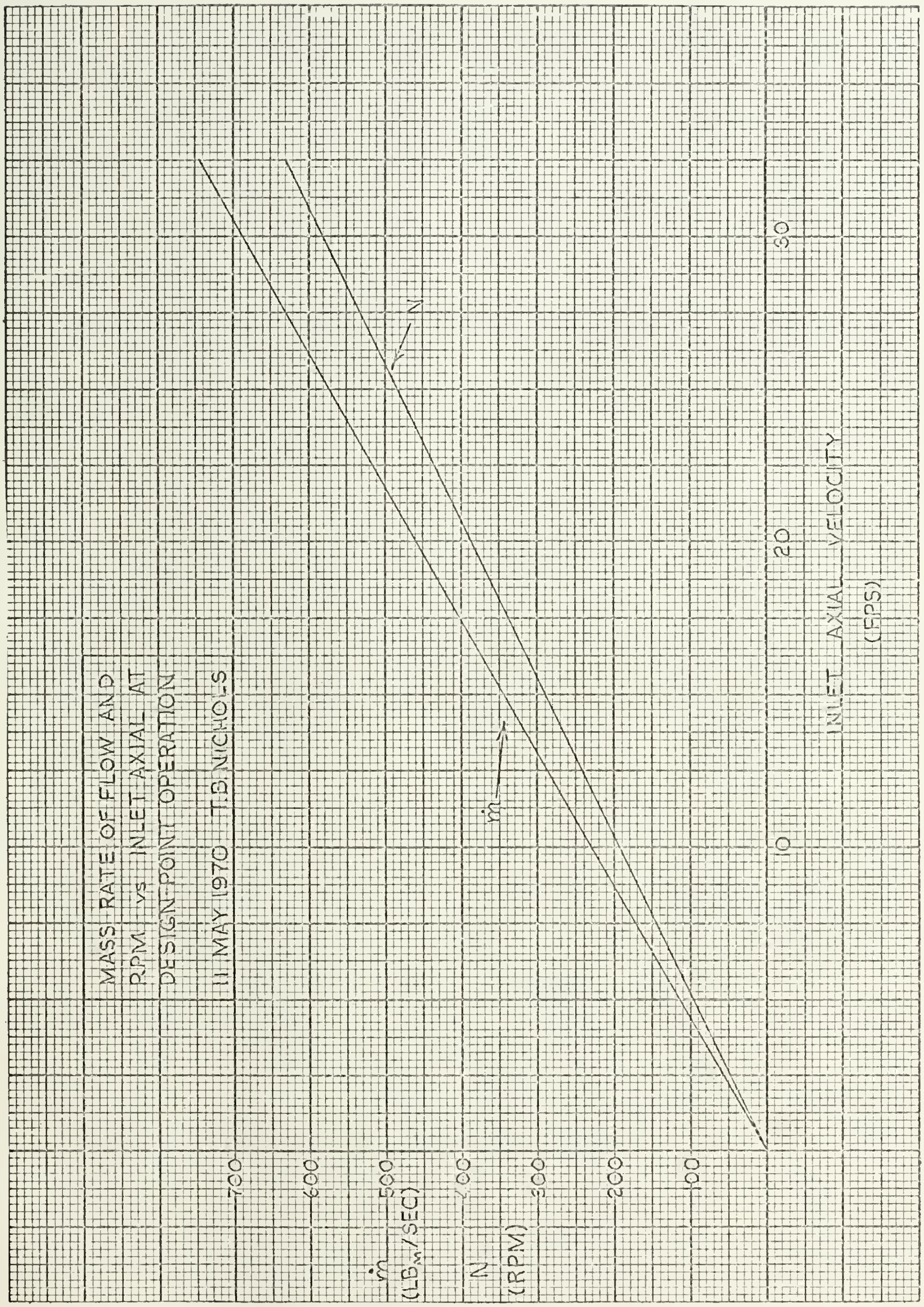
$$K_3 = 0.814$$

$$K_4 = \frac{N}{C_x} = \frac{630}{32.7}$$

$$K_4 = 19.25$$

C_x	$\dot{m}$	N	T	HP
1	23.6	19.3	0.814	0.000475
5	118.0	96.3	20.35	0.0594
10	236.0	193.0	81.40	0.475
15	354.0	289.0	183.0	1.603
20	472.0	385.0	325.0	3.79
25	590.0	482.0	508.0	7.43
30	707.0	577.0	733.0	12.80
32.7	744.0	630.0	870.0	16.61


TORQUE AND HORSEPOWER
vs
INLET AXIAL VELOCITY
AT DESIGN-POINT OPERATION
11 MAY 1970 T.B. NICHOLS
TORQUE
(FT-LBS)
100
200
300
400
500
600
700
800
HORSE-
POWER
5.0
10.0
15.0
20.0
INLET AXIAL VELOCITY
(FPS)
10
20
30



MASS RATE OF FLOW AND
RPM vs INLET AXIAL AT
DESIGN POINT OPERATION
11 MAY 1970 T.B. NICHOLS
700
600
500
400
300
200
100
$\dot{m}$
(LB_M/SEC)
N
(RPM)
10
20
30
INLET AXIAL VELOCITY
(FPS)
$\dot{m}$
N

Thesis
N4864
118375
Nichols
Visualization of secondary-flow phenomena in cooled gas-turbine blades.